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MATHEMATICAL ANALYSIS OF AIRCRAFT INTERCOOLER DESIGN

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SUMMARY

A mathematical analysis has been made to show the method of obtaining the dimensions of the intercooler that will use the least total power for a given set of design conditions.

The results of this analysis have been used in a sample calculation and, on the basis of this calculation, a new intercooler arrangement is suggested. Because the length of the two air passages of the new arrangement is short in comparison with the third dimension, the height of the intercooler, this intercooler arrangement has unusual dimensions. These dimensions give the proposed intercooler arrangement an advantage over one of usual dimensions because less total power will be consumed by the intercooler, the weight and the volume of the intercooler will be smaller, and the pressure drop of both the engine air and the cooling air in passing through the intercooler will be lower.

INTRODUCTION

The design of an optimum intercooler involves the solution of a problem having many variable factors. A whole series of intercoolers can be designed that will accomplish a required transfer of heat with certain specified temperature differences between cooling air and heated, or engine, air. These intercoolers, all of which meet the required conditions of heat transfer, will vary widely in volume, power consumed by the intercooler, linear dimensions of the intercooler, pressure drop of the engine air, and pressure drop of the cooling air in passing through the intercooler. Obviously, some criterion must be established by which the best intercooler of the series can be selected. Such a criterion should consider all the significant variables.

The total power consumed by the intercooler is used as the design criterion in this analysis. This total power includes both the power used in forcing the engine air and the cooling air through the intercooler, and the power required to transport the weight of the intercooler. For a stationary intercooler, in which the power for transporting the intercooler is not included in the calculations, the size of the optimum intercooler approaches infinity as the power approaches zero. The criterion for the design of a stationary intercooler would, therefore, not be the minimum power consumed by the intercooler but would be a factor like its initial cost.

For an aircraft intercooler, however, the total power consumed is an expedient criterion because the power required for the transportation of the intercooler is proportional to the weight of the intercooler, which is, in turn, proportional to the intercooler volume. On the basis of the criterion selected, an extremely large intercooler would be the most efficient if only the power required to force the air through the intercooler were considered; whereas, an extremely small intercooler would be the best if only the power required to transport the intercooler were considered. An intercooler of an optimum size is certain to exist for the minimum total-power consumption.

The design conditions of the intercooler will include values for: the total mass flow of engine air, the inlet temperatures of both cooling and engine air, the required temperature of the engine air at the outlet from the intercooler, the characteristics of the airplane in which the intercooler is to be installed, and the physical characteristics of the cooling air.

The type of intercooler core to be used can be chosen from considerations of construction and of known principles of heat transfer. Experience indicates that the type of construction used should be such that both the heat-transfer coefficient and the ratio of the heat-transfer coefficient to the friction factor are large. This requirement means that Reynolds' analogy should be approached as closely as possible. Any turbulence-producing device that is introduced in an effort to get high values of the heat-transfer coefficient will disproportionately increase the power consumption and will be disadvantageous from considerations of power consumption. Small, smooth passageway construction offers a means of achieving the desired conditions.

Whether counterflow or crossflow of the air streams is to be used is virtually determined by manifolding considerations. The counterflow type of construction has a slight advantage over the crossflow type on the basis of the quantity of heat transferred, all other factors being equal. In a practicable case this advantage might amount to 10 to 15 percent. The greater ease of manifolding a crossflow intercooler as compared with that of manifolding one of the counterflow type is usually considered of sufficient importance to outweigh the advantage of a greater heat transfer in the counterflow intercooler. A crossflow intercooler has been investigated and the results are presented in the present paper.

The type of intercooler assumed for the purpose of this analysis is shown in figure 1. Symbols are defined in the next section. The passageways are assumed to be smooth and the air flow is assumed to be turbulent. Experience indicates that the Reynolds number of the air flow and the initial turbulence will insure turbulent flow.

Heat-transfer-coefficient and friction-factor data for round pipes are applied to the rectangular passages. McAdams (reference 1, p. 117) states that this application is permissible with turbulent flow, provided that the hydraulic diameter of the passageway is used for the tube diameter in all calculations.

The fin effectiveness is assumed constant. This value obviously varies, but numerous calculations on intercoolers of this finned type have indicated values of 93 to 97 percent for fin effectiveness. A fin effectiveness of 95 percent will be close to the actual value for usable intercoolers.

End losses are disregarded for simplification. Considerations of the tube dimensions and the friction factor indicate a maximum error of about 5 percent in power consumption in the usable range as a result of disregarding the end losses.

NOTATION

The units given are the ones used in this paper. Any self-consistent system of units may be used.

A free area for passage of air, square foot

c_p specific heat of air at constant pressure, Btu per slug per degree Fahrenheit

C_D drag coefficient of airplane in cruising condition

C_L lift coefficient of airplane in cruising condition

D hydraulic diameter of air passage

$$\left(\frac{4 \text{ cross-sectional area}}{\text{wetted perimeter}}\right), \text{ foot}$$

f friction factor $\left(\frac{\Delta p}{\frac{1}{2}\rho V^2}\,\frac{D}{4L}\right)$

f' fin effectiveness

h_t over-all heat-transfer coefficient from fluid to fluid based on dividing-plate area, Btu per second per square foot per degree Fahrenheit

h_s surface heat-transfer coefficient

h_c heat-transfer coefficient from cooling air to dividing plate

h_e heat-transfer coefficient from engine air to dividing plate

H height of intercooler, feet

L_c length of cooling-air passage, feet

L_e length of engine-air passage, feet

M mass flow of air, slugs per second

Δp over-all pressure drop through intercooler, pounds per square foot

P_e, P_c total power required to force engine air or cooling air through intercooler, foot-pounds per second

P_w total power required to transport intercooler, foot-pounds per second

P_t total power consumed by intercooler, foot-pounds per second

Q volume flow of air, cubic feet per second

s spacing between fins, feet

S total dividing-plate area, square feet

t_f fin thickness, feet

t_p dividing-plate thickness, feet

T temperature of the engine air, degrees Fahrenheit

T' temperature of the cooling air, degrees Fahrenheit

V average velocity of air flow through intercooler, feet per second

V_o velocity of airplane, feet per second

w width of fins, feet

W weight of intercooler, pounds

ϵ factor to take care of weight of intercooler mounting, etc.

ζ mean over-all temperature difference between engine air and cooling air for crossflow divided by $T_i - T'_i$, given by the empirical relation (equation (13)) developed for this paper

$\eta = \dfrac{T'_o - T'_i}{T_i - T'_i}$ relates to cooling air

$\xi = \dfrac{T_i - T_o}{T_i - T'_i}$ relates to engine air

μ coefficient of viscosity of air, slugs per foot-second

ρ mass density of air, slugs per cubic foot

$$\left.\begin{array}{l} \phi_1 \ \cdots \ \phi_n \\ \theta_1 \ \cdots \ \theta_n \\ C_1 \ \cdots \ C_n \\ \delta_1 \ \cdots \ \delta_n \\ \omega_1 \ \cdots \ \omega_n \end{array}\right\} \text{constants}$$

Subscripts:

c refers to cooling air

e refers to engine air

i refers to inlet air

o refers to outlet air, datum

Definition of Functions

$$C_1 = \frac{1}{2}\,\frac{w}{(w + t_p)}\,\frac{s}{(s + t_f)} \qquad \text{(See fig. 1.)}$$

C_2 = weight per unit volume of intercooler, pounds per cubic foot

$$C_3 = \frac{0.4\ \delta_2\omega_3}{\delta_3\ \omega_2}$$

$$C_4 = \left(\frac{2.5\ \delta_3\ \omega_1}{C_3^{\,2.6}\ \delta_1\omega_3}\right)^{\frac{1}{5.76}}$$

$$\phi_1 = \frac{0.098\ \mu_c^{\,0.2}}{C_1^{\,1.8}\rho_c^{\,2.0}D^{1.2}}$$

$$\phi_2 = \frac{0.098\ \mu_e^{\,0.2}}{C_1^{\,1.8}\rho_e^{\,2.0}D^{1.2}}$$

$$\phi_3 = C_2 \epsilon V_0 \frac{C_D}{C_L}$$

$$\theta_2 = \left(\frac{s + t_f}{wf' + s}\right)\left(\frac{D^{0.2}\ C_1^{0.8}}{0.0245\ c_p \mu_c^{0.2}}\right)$$

$$\theta_3 = \left(\frac{s + t_f}{wf' + s}\right)\left(\frac{D^{0.2}\ C_1^{0.8}}{0.0245\ c_p \mu_e^{0.2}}\right)$$

$$\theta_4 = \theta_2 M_e$$

$$\theta_5 = \theta_3 M_e^{0.2}$$

$$\theta_7 = \frac{1}{(w + t_p)\ c_p \log_e \left(\frac{1}{1 - \xi}\right)}$$

$$\theta_8 = \frac{0.46}{\sqrt{1 - \xi}}$$

$$\omega_1 = \phi_1\ M_c^{2.8} \qquad \delta_1 = \frac{\theta_4}{M_c^{0.8}}$$

$$\omega_2 = \phi_2\ M_e^{2.8} \qquad \delta_2 = \theta_5$$

$$\omega_3 = \phi_3 \qquad \delta_3 = \theta_7 \left(1 - \frac{M_e}{M_c}\xi\right)^{\theta_8}$$

ANALYSIS

The various considerations stated in the Introduction make it possible to assign values to all of the variables except four: (1) total mass flow of cooling air; (2) length of air passage in the direction of cooling-air flow; (3) length of air passage in the direction of engine-air flow; and (4) the third dimension, or height, of the intercooler. The purpose of this paper is to show how to determine the values of these four variables so that the total power consumed by the intercooler shall be a minimum.

The general plan of this analysis is to obtain an expression for the total power consumed in terms of the four variables M_c, L_c, L_e, and H, and then to minimize this power, subject to the relation between the variables that is imposed by the required conditions of heat transfer. An expression will first be obtained for the power used in forcing cooling air through the intercooler. The equation for the friction factor in terms of the Reynolds number was obtained from reference 1 (p. 111), and is

$$f = \frac{\Delta p}{\frac{1}{2}\rho V^2}\,\frac{D}{4L} = \frac{0.049}{\left(\frac{\rho VD}{\mu}\right)^{0.2}} \tag{1}$$

from which

$$\Delta p_c = \frac{0.098\ \mu_c^{0.2}\ L_c\ \rho_c^{0.8}\ V_c^{1.8}}{D_c^{1.2}} \tag{2}$$

The free area for the passage of cooling air through the intercooler is

$$A_c = C_1\ L_e\ H \tag{3}$$

The velocity of flow of cooling air through the intercooler may be written as

$$V_c = \frac{Q_c}{A_c} = \frac{\frac{M_c}{\rho_c}}{C_1\ L_e\ H} \tag{4}$$

From equations (2) and (4) may be obtained the expression for the power required to force cooling air through the intercooler:

$$P_c = Q_c\Delta p_c = \frac{M_c^{2.8}\ L_c}{L_e^{1.8}\ H^{1.8}}\left(\frac{0.098\ \mu_c^{0.2}}{C_1^{1.8}\ \rho_c^{2.0}\ D^{1.2}}\right) \tag{5}$$

$$P_c = \phi_1\ \frac{M_c^{2.8}\ L_c}{L_e^{1.8}\ H^{1.8}} \tag{6}$$

and, similarly,

$$P_e = \phi_2 \frac{M_e^{2.8} L_e}{L_c^{1.8} H^{1.8}} \tag{7}$$

The power required to transport the weight of the intercooler is given by

$$P_w = W \epsilon V_o \frac{C_D}{C_L} \tag{8}$$

and the weight of the intercooler under consideration is given by

$$W = C_2 HL_cL_e \tag{9}$$

where C_2 is the weight per unit volume of the intercooler.

From equations (8) and (9),

$$P_w = \left(C_2 \epsilon V_o \frac{C_D}{C_L}\right) HL_cL_e = \phi_3 HL_cL_e \tag{10}$$

The total power consumed is given by the sum of equations (6), (7), and (10) as

$$P_t = \phi_1 \frac{M_c^{2.8} L_c}{L_e^{1.8} H^{1.8}} + \phi_2 \frac{M_e^{2.8} L_e}{L_c^{1.8} H^{1.8}} + \phi_3 HL_cL_e \tag{11}$$

The relation between the variables imposed by the required conditions of heat transfer can be developed directly from the expression that equates the total heat given up by the engine air to the total heat transferred through the dividing plates,

$$M_e c_p \xi = h_t S \zeta \tag{12}$$

Nusselt (reference 2) has given a theoretical relation for ζ in terms of ξ and η, but the relation is too complicated for general use. Nusselt has also given a table of values of ζ, but, in order to use ζ analytically, an empirical relation for ζ has been developed from Nusselt's results that gives values correct to within 1 percent, provided that neither ξ nor η exceeds 0.7.

For larger values of ξ and η, the error approaches 5 percent. The empirical relation is

$$\zeta = \frac{\xi(1-\eta)^{\sqrt[0.46]{1-\xi}}}{\log_e\left(\frac{1}{1-\xi}\right)} = \frac{\xi\left(1-\frac{M_c}{M_c}\xi\right)^{\sqrt[0.46]{1-\xi}}}{\log_e\left(\frac{1}{1-\xi}\right)} \tag{13}$$

The total dividing-plate area S is

$$S = \frac{H}{w+t_p} L_c L_o \tag{14}$$

An expression for the local surface heat-transfer coefficient is given in reference 1 (p. 173) as

$$h_s = 0.0245 \frac{c_p\, \mu^{0.2}\, \rho^{0.8}\, V^{0.8}}{D^{0.2}} \tag{15}$$

An examination of figure 1 shows that the heat-transfer coefficient on the cooling-air side, based on dividing-plate area, is

$$h_c = \frac{wf' + s}{s + t_f} h_s \tag{16}$$

Substituting equations (4) and (15) in equation (16) and inverting,

$$\frac{1}{h_c} = \left(\frac{s+t_f}{wf'+s} \frac{D_c^{\ 0.2}\ C_1^{\ 0.8}}{0.0245\ c_p\mu_c^{\ 0.2}}\right)\frac{L_e^{\ 0.8}\ H^{0.8}}{M_c^{\ 0.8}} = \theta_2 \frac{L_e^{\ 0.8}\ H^{0.8}}{M_c^{\ 0.8}} \tag{17}$$

and, similarly,

$$\frac{1}{h_e} = \theta_3 \frac{L_c^{\ 0.8}\ H^{0.8}}{M_e^{\ 0.8}} \tag{18}$$

The equation giving the over-all heat-transfer coefficient when two surface coefficients are acting in series is

$$\frac{1}{h_t} = \frac{1}{h_c} + \frac{1}{h_e} \tag{19}$$

and, from equation (12),

$$\frac{1}{h_t} = \frac{S\,\zeta}{M_e c_p\,\xi}$$

so that

$$\frac{S\,\zeta}{M_e c_p \xi} = \frac{1}{h_c} + \frac{1}{h_e} \tag{20}$$

Substituting equations (13), (14), (17), and (18) into equation (20) gives

$$\frac{\theta_4\ L_e^{0.8}}{M_c^{0.8}} + \theta_5\ L_c^{0.8} - \theta_7\ H^{0.2}\ L_c L_e \left(1 - \frac{M_e}{M_c}\xi\right)^{\theta_8} = 0 \tag{21}$$

Equation (21) is the required expression for the relation between the variables if the stipulated conditions of heat transfer are to be met and is the constraint to be used when minimizing equation (11).

The problem now is to obtain a minimum value for the total power consumption as given by equation (11), subject to the requirement that the variables shall at all times satisfy equation (21). This minimum value is obtained by means of Lagrange's method of undetermined multipliers (reference 3, p. 120).

In order to obtain conditions for the maximum or the minimum value of the function $P_t = F(M_c, L_c, L_e, H)$ where the variables are connected by a relation $G(M_c, L_c, L_e, H) = 0$, the operations indicated in the following equations are performed:

$$\left.\begin{aligned}
\frac{\partial F}{\partial M_c} + \lambda \frac{\partial G}{\partial M_c} &= 0 \\
\frac{\partial F}{\partial L_c} + \lambda \frac{\partial G}{\partial L_c} &= 0 \\
\frac{\partial F}{\partial L_e} + \lambda \frac{\partial G}{\partial L_e} &= 0 \\
\frac{\partial F}{\partial H} + \lambda \frac{\partial G}{\partial H} &= 0
\end{aligned}\right\} \qquad (22)$$

where λ is an unknown multiplier.

Equations (22) taken with equation (21) give five simultaneous equations in the four variables and λ, and the solution of these equations gives the value of the variables for minimum or maximum power.

This method of solution was first applied to equation (11) subject to the constraint imposed by equation (21). The only solution obtained was $M_c = \infty$ for minimum power, which indicated an intercooler only of academic interest outside the region where the assumptions are valid. A value was assigned to M_c in order to obtain solutions for the other three variables. The assigned value of M_c may be included in one of the constants in each equation in which it occurs, and equations (11) and (21) reduce to

$$P_t = F(L_c, L_e, H) = \omega_1 \frac{L_c}{L_e^{1.8} H^{1.8}} + \omega_2 \frac{L_e}{L_c^{1.8} H^{1.8}} + \omega_3 \; H L_c L_e \qquad (23)$$

$$G(L_c,\ L_e,\ H) = \delta_1 L_e^{0.8} + \delta_2 L_c^{0.8} - \delta_3 H^{0.2} \, L_c \, L_e = 0 \qquad (24)$$

If the operations indicated in equation (22) are performed in equations (23) and (24), three equations result:

$$\left.\begin{aligned}
&\frac{\omega_1}{L_e^{1.8}\ H^{1.8}} - \frac{1.8\ \omega_2 L_e}{L_c^{2.8}\ H^{1.8}} + \omega_3 H L_e + \lambda\left(\frac{0.8\ \delta_2}{L_c^{0.2}} - \delta_3 H^{0.2}\ L_e\right) = 0 \\
&\frac{-\ 1.8\ \omega_1\ L_c}{L_e^{2.8}\ H^{1.8}} + \frac{\omega_2}{L_c^{1.8}\ H^{1.8}} + \omega_3 H L_c + \lambda\left(\frac{0.8\ \delta_1}{L_e^{0.2}} - \delta_3 H^{0.2}\ L_c\right) = 0 \\
&\frac{-\ 1.8\ \omega_1\ L_c}{L_e^{1.8}\ H^{2.8}} - \frac{1.8\ \omega_2 L_e}{L_c^{1.8}\ H^{2.8}} + \omega_3\ L_c L_e + \lambda\left(-\ 0.2\ \frac{\delta_3\ L_c L_e}{H^{0.8}}\right) = 0
\end{aligned}\right\} \quad (25)$$

These three equations, together with equation (24), form a set of four simultaneous equations in the four variables L_c, L_e, H, and λ. The exact solutions for these equations have been obtained, and are

$$H = \left(\frac{0.4\ \omega_3\ C_3\ \ C_4^{18/5}}{\dfrac{\omega_1}{C_3^{9/5}\ \ C_4^{92/25}} + \omega_2\, C_3\, C_4^{4/5}}\right)^{4/7} \quad (26)$$

$$L_c = C_4\ H^{-13/8} \quad (27)$$

$$L_e = C_3\ (L_c H)^{13/5} \quad (28)$$

When these three solutions were obtained, values for λ and P_t were also obtained:

$$\lambda = \frac{1.4\ \omega_3\ H^{4/5}}{\delta_3} \quad (29)$$

and

$$P_t = 1.4\ P_w \quad (30)$$

Equation (30) serves as a check on the accuracy of calculation. The calculations made for this paper checked to five or six significant figures in equation (30). A more complete check on the calculations is the comparison of the value of Δp_c, as calculated from equation (2), with the value of Δp_c calculated from $\Delta p_c = \frac{P_c}{Q_c}$ (from equation (5)).

EXAMPLE

For illustration of the results to be derived from this analysis, calculations of optimum intercooler dimensions are made on the basis of the following assumptions:

1. The brake horsepower of the engine is 1000.

2. The engine uses 6600 pounds of air per hour, or 0.0569 slug per second.

3. $T_i = 280^\circ$ F, $T_o = 80^\circ$ F, $T'_i = -30^\circ$ F.

4. The airplane is operating at the rated height of the engine, which is 25,000 feet.

5. V_o, the flight velocity of the airplane, is 300 miles per hour or 440 feet per second.

6. $\epsilon \frac{C_D}{C_L} = 0.075$.

7. The intercooler is made of copper (555 lb per cu ft); $s = 1/16$ inch; $w = 1/2$ inch; $t_f = 0.005$ inch; $t_p = 0.010$ inch.

Fluid Constants Used

	Engine air	Cooling air
Density, slugs/cu ft	0.001965 at 180° F	0.000907 at 20° F
Viscosity, slugs/ft-sec	0.443×10^{-6} at 180° F	0.355×10^{-6} at 20° F
Thermal conductivity, Btu/sec/ft²/°F/ft	4.19×10^{-6} at 140° F	3.61×10^{-6} at 40° F
Specific heat, Btu/slug/°F	7.73	7.73
Pressure, in. Hg	30.5	11.1

The results of the calculations made are shown in table I and in figure 2.

Pumping efficiencies have not been included in these computations. If desired, the pumping efficiencies may be included in the constants ω_1 and ω_2 by dividing the constant by the efficiency.

TABLE I

Results of Sample Calculation

M_c	H (ft)	L_c (ft)	L_e (ft)	Intercooler volume (cu ft)	Δp_c (lb/sq ft)	Δp_e (lb/sq ft)	P_t (hp)
1.00	69.91	0.018	0.470	0.583	0.208	5.57	2.50
.30	19.72	.063	.497	.617	.745	5.89	2.66
.15	8.64	.144	.568	.704	1.700	6.72	3.03
.05	1.29	.964	1.270	1.576	11.41	15.04	6.78
.04	.42	2.974	3.136	3.890	35.24	37.14	16.75

DISCUSSION AND CONCLUSIONS

It is evident that the foregoing analysis cannot be applied without some alteration to other types of intercooler than the crossflow one assumed. The general appearance of curves like those shown in figure 2 will be similar for all types of crossflow intercoolers, so that the analysis presented in this paper should serve as a guide in the testing of all similar types.

In view of the simplifying assumptions made for this analysis and the number of sources from which the data were drawn, it would not be surprising if the values of power and the intercooler dimensions calculated on the basis of the analysis should be somewhat in error but the trends shown in figure 2 should be correct.

An examination of figure 2 shows the advisability of using large values of mass flow of cooling air. Increasing M_c reduces the total power consumed, the volume and weight of the intercooler, and the pressure drop of both engine and cooling air. The objections to using large values of M_c are: (1) More air must be taken into the airplane from the main air stream, and (2) the intercooler assumes very elongated proportions. As regards the first objection, even though more air is passed through the airplane with an increase in M_c, the energy loss is much lower, so that a reasonable increase in M_c may be tolerated. It can be seen from figure 2 that, at $M_c = 0.10$, a large part of the advantage to be gained from a large value of mass flow of cooling air has been obtained. The second objection might be overcome if the elongated intercooler is used in an arrangement similar to the one shown in figure 3. It seems as though an arrangement such as this one can be made with no more over-all volume than in an optimum intercooler, with its manifolding, using small cooling air flow. Other arrangements for using an elongated intercooler core may be better; figure 3 is only one suggestion.

In previous calculations for the design of an intercooler complying with the conditions specified in the example given in the paper, an attempt was made to assign values to the variables by inspection and successive approximations. The lowest value of the total power consumed that was obtained from any of these previous calculations was about 8 horsepower; whereas, the present calculations immediately indicate an intercooler that will consume only 3 horsepower. The volume and the weight of the optimum intercooler also are smaller in the present analysis.

Langley Memorial Aeronautical Laboratory,
National Advisory Committee for Aeronautics,
Langley Field, Va., August 27, 1940.

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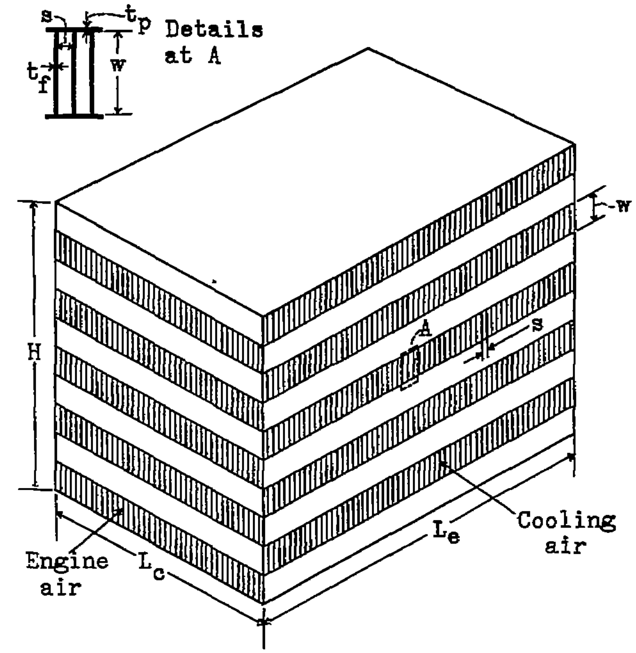

Figure 1 .- Type of intercooler considered.

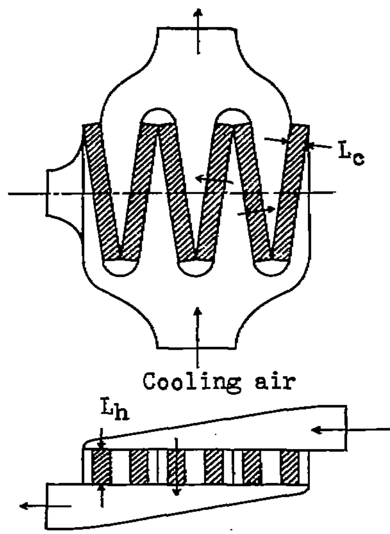

Figure 3.- Suggested type of intercooler.

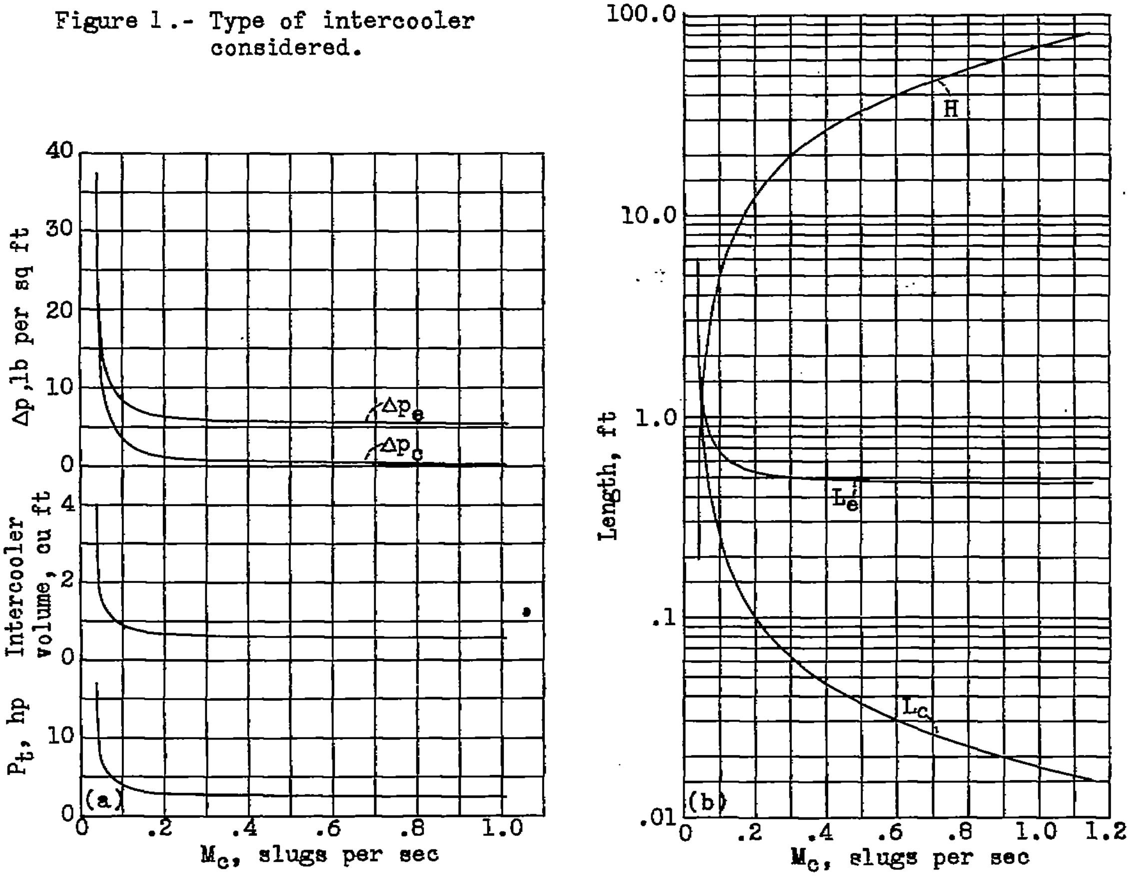

Figure 2.- Variation with M_c of optimum values of intercooler parameters. (Taken from results of sample calculation. See table 1.)
(a) Variation of horsepower consumed, intercooler volume, and pressure drop.
(b) Variation of intercooler dimensions.